When compressor seals act like bearings

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This case history documents a phenomenon occasionally found in machinery utilizing bushing-type oil film process gas seals. By design, this type of seal is intended to float radially (Figure 1). In certain applications, however, the seal may lock up and behave as an externally-lubricated bearing which can unload the tilting pad bearings and change the balance resonance speeds significantly. In this case, the first balance resonance speed is at the midpoint of the governor speed range for this machine.

The data for this analysis was recorded at a plant where several compressors were monitored for vibration during a startup. The data presented was collected during two consecutive starts of one of the compressors.

The machine is a barrel type centrifugal compressor designed to compress hydrogen gas for recycle and quench processing. The compressor is directly driven through a gear coupling by a steam turbine which has a governor speed range of 7,716 rpm to 11,025 rpm (Figure 2). The rotating elements consist of a shaft, three impellers and a balance piston. Oil film seals are used to prevent process gas leakage to the atmosphere. The compressor is rated at 468 ACFM with a suction pressure of 2,855 psig (201 bar), a discharge pressure of 3,125 psig (220 bar) and an outlet temperature of 145°F (63°C) at a speed of 9,000 rpm.

History

The first startup after overhaul occurred on March 15. Bently Nevada's Machinery Diagnostic Services group was on-site to monitor the startup. While the compressor was operating at 8,900 rpm, several Bently Nevada 7200 vibration monitors were in alarm above 1.75 mils (44 μm) pp, and the inboard compressor bearing was just below the shutdown level of 2 mils (51 μm) pp. Surprisingly, the data indicated that the compressor rotor did not encounter the first balance resonance (critical) during startup. This was attributed to a possible rub creating additional stiffness in the rotor system.

Normally, a machine rotor's balance resonance speeds for each mode are determined by the rotor assembly's

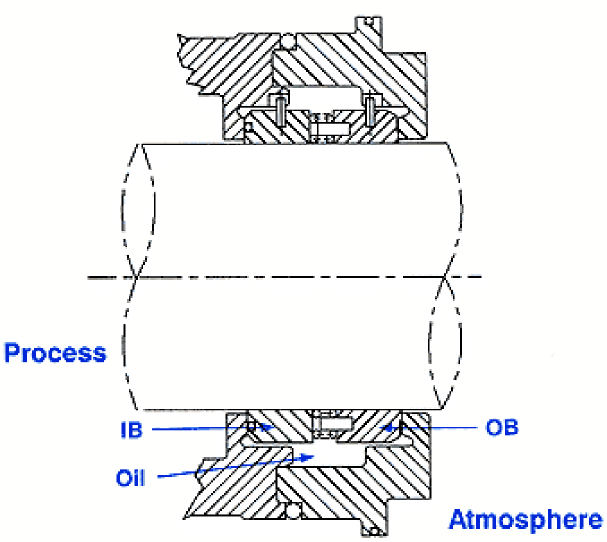

Figure 1
Typical Floating Seal Ring Arrangement

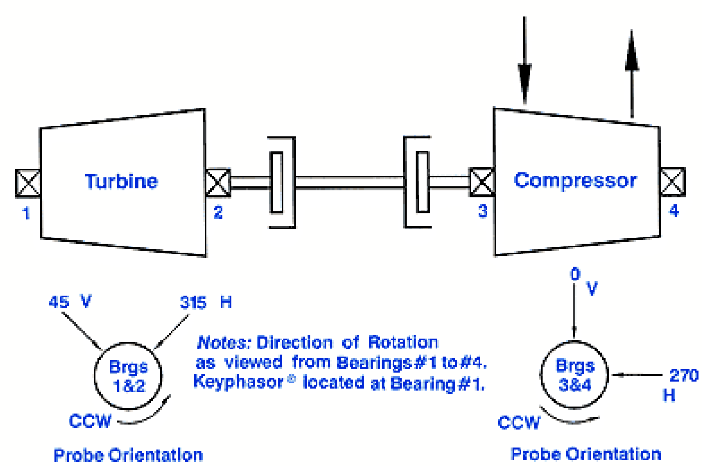

Figure 2
Machine Train Diagram

undamped natural frequency as given by the following equation:

$$\omega_n = \sqrt{\frac{K}{M}}$$

Where:

ω_n = natural frequency
K = rotor system modal stiffness
M = rotor modal mass

The balance resonance is affected by changes in either rotor mass or rotor system stiffness. Rotor mass does not change significantly during machine operation; however, the rotor system stiffness may change considerably due to the presence of a rub, oil temperature variance, or other conditions. For additional information on dynamic stiffness, please see Reference 1.

On Sunday, March 17, the vibration monitor for the inboard compressor bearing tripped the machine off-line when vibration exceeded 2 mils (51 μm) pp during a small speed increase. An ADRE® 3 System was used to generate the shutdown Polar plot from the inboard end of the compressor rotor (Figure 3). This plot suggested that the rotor was operating near its balance resonance (critical). The restart Polar plot (Figure 4) was in agreement with the shutdown data. Since the machine was not operated at higher speed immediately following restart, the speed of maximum vibration amplitude was extrapolated to be around 10,000 cpm.

The steady state vibration amplitudes after restart were similar to those experienced prior to the trip shutdown, with approximately 2 mils (51 μm) pp at the compressor inboard bearing. At this time, the Maintenance Engineering Department was advised that the compressor should be shut down for investigation of the cause of the increased rotor system stiffness that seemed responsible for elevating the balance resonance from the expected 5,700 cpm (operating manual) to above 10,000 cpm. The condition was also thought to be aggravated by the hot alignment condition of the machine based on shaft centerline data acquired during the trip and the subsequent restart. The vibration amplitudes were minimized by limiting the operating speed to less than 9,000 rpm.

The compressor performance was good, and the vibration condition had not changed for several days. As a precaution, data was being digitally recorded on the ADRE® 3 System.

On March 23, a sudden change in amplitude and phase was noted at both compressor bearings. The direct vibration amplitude at the compressor inboard bearing decreased from 2 mils (51 μm) pp to 1 mil (25 μm) pp. A 180° phase shift at the inboard bearing occurred 2.5 hours after the drop in magnitude (Figure 5). The outboard end of the rotor had very low vibration amplitudes; however, the small increase in amplitude and a 60° phase shift occurred simultaneously.

Although this appeared to be an improvement in the operating condition of the machine, the plant was advised that the balance resonance, which had been extrapolated to occur at about 10,000 cpm, had passed down through the operating speed range due to a change in the rotor system stiffness. Something in the rotor system had changed to affect the stiffness.

By this time, the compressor and the related refinery process had reached steady state production, allowing additional time to review data and theorize. As the operation continued, hydrogen leaks developed at both ends of the compressor. Adjustments were made to the lubrication and seal oil system that eliminated the leakage problem. Up to this point, incorrect alignment had been considered the most likely cause of the increased system stiffness. However, when seal and lubrication system measurements were completed, it was obvious that the seals were damaged.►

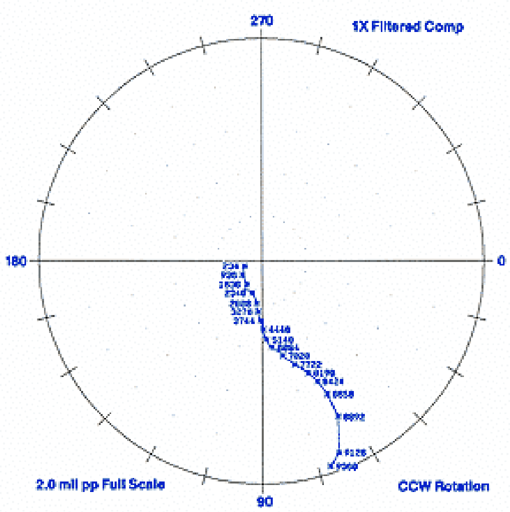

Figure 3
Inboard end of the compressor rotor.
Polar plot showing shutdown data

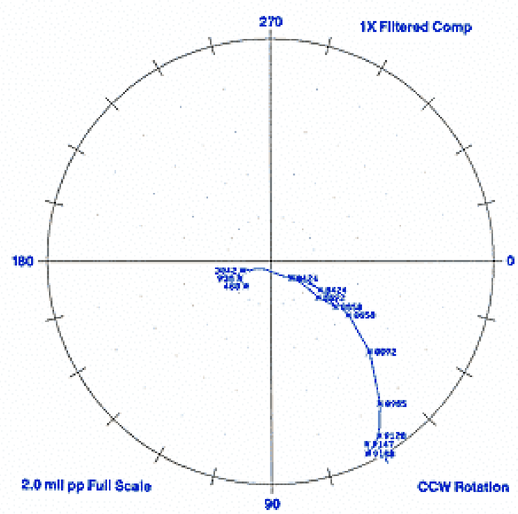

Figure 4
Inboard end of the compressor rotor.
Polar plot showing restart data

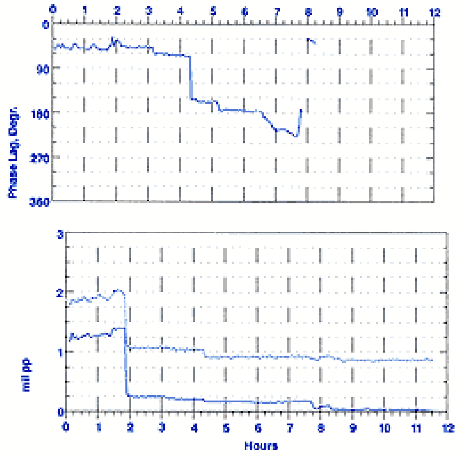

Figure 5
Inboard end of the compressor rotor. Steady state 1X data showing a 180° phase shift at the inboard bearing

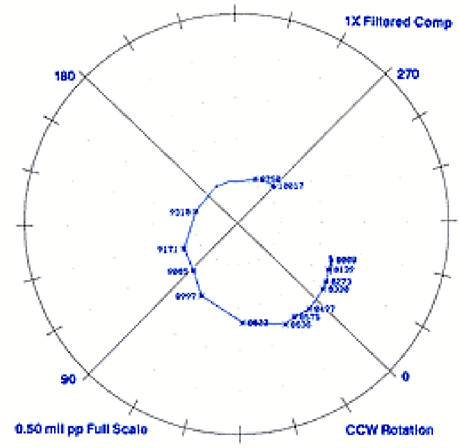

Figure 6
Outboard end of the compressor rotor. Overspeed data

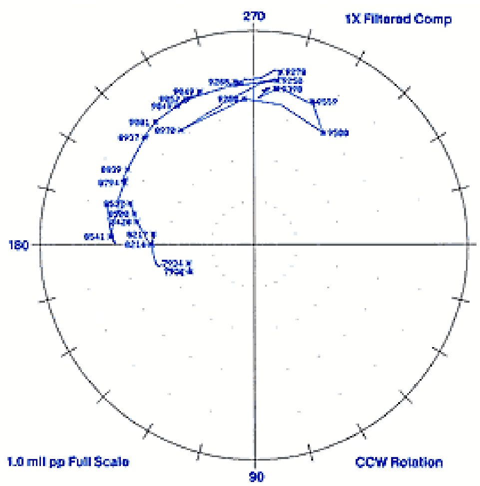

Figure 7
Inboard end of the compressor rotor.
Polar plot showing restart data

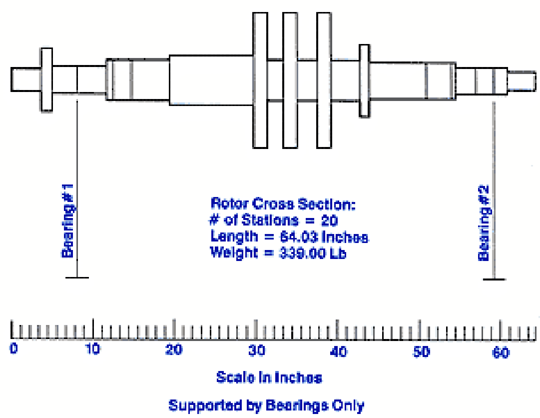

Figure 8
Rotor model cross section.

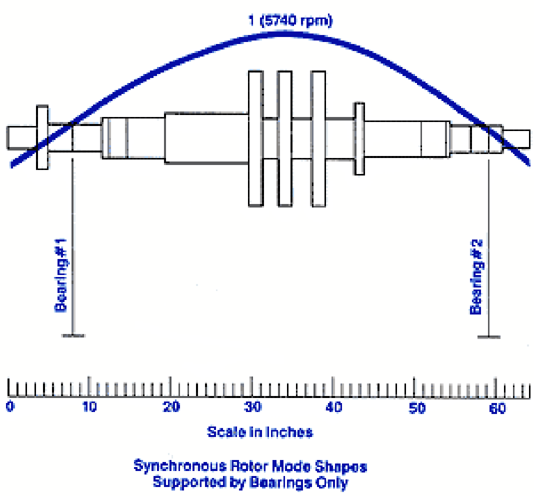

Figure 9
Baseline rotor model supported by two bearings

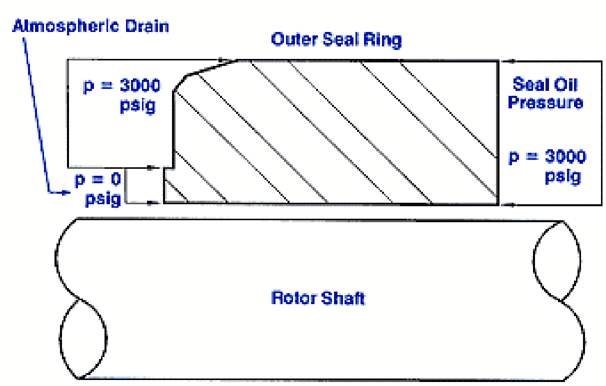

Figure 10
Diagram of outboard seal ring

These measurements indicated that the seal oil flow was much higher than normal, presumably due to an increase in the seal clearance. In this condition, very little flow margin remained in the seal oil pumping system.

To determine the actual range of the balance resonance, a variable speed test was conducted on April 5. The result of that test is shown in Figure 6. This range was found to be 8,800 to 9,300 cpm, well above the 5,700 cpm specified by the manufacturer.

The compressor was shut down on April 28 and found to have a scored rotor in the vicinity of both inboard and outboard seal rings. The inside of the seal rings was also found to be damaged from contact with the shaft. As a precaution, a new rotor, seals, and bearings were installed. After shutdown, minor adjustments were made in the compressor-to-turbine alignment based on Essinger bar readings taken just prior to shutdown.

The compressor was restarted on May 5. The first balance resonance was again determined to be between 8,800 and 9,200 cpm by variable speed testing. The resultant Polar plot is shown in Figure 7. Except for the elevated balance resonance, there were no other significant indicators of machinery malfunction. It was again noted that the compressor's first balance resonance coincided with the normal operating speed range for this process.

Analysis

Modal analysis was performed on the compressor rotor to gain a better understanding of the rotor's behavior. Figure 8 is a cross section of the rotor model. From left to right are the thrust collar, the three impellers, and the balance piston.

The baseline model is supported by two bearings 51 inches (1.3 metres) from center to center. The model predicted a first balance resonance speed of 5,740 rpm (Figure 9) which is in good agreement with the 5,700 rpm published in the operating manual. The model was constructed using measured dimensions from a rotor assembly. The shaft density is .286 lb/in^3 (7.92 g/cm^3) and an impeller density of 0.12 lb/in^3 (3.32 g/cm^3) was used to model the impeller as a solid disk. The bearing stiffness was determined by trial and error after the geometry and weight were established. The stiffness of the bearings used in this model is 1.0×10^6 lb/in (1.79×10^8 g/cm) which is reasonable for a tilting pad bearing with load-on-pad configuration. Since this model corresponds to the actual rotor weight, size, and resonant speed (predicted), it is judged to be suitable for demonstrating how the seal components might affect the rotor dynamics.

First, it is important to understand how the seal can be involved with the shaft dynamics. The process (hydrogen) is inboard of the seal assemblies (Figure 1). There are also labyrinth seals just inboard of each floating seal ring assembly. Labyrinth seals are not shown and are not considered in this discussion. Also, aerodynamic seals are not considered here.

The process gas pressure is approximately 3,000 psig (211 bar). Oil is supplied between the seal rings at 10-15 psi (68,948 to 103,422 N/m^2) higher than the process gas pressure. Seal oil flows constantly across the laminar clearance between the shaft and both seal rings thus eliminating gas leakage. The inner seal ring is pressure-balanced by the nearly equal process gas and oil pressures acting on either side of it. The outer seal is not so fortunate.

The oil pressure acting on the inner face of the outboard seal ring is approximately 3,000 psi (211 bar). The outer face of the outboard seal ring is lapped and forms a seal with the lapped face of the seal housing. The lapped seal area is not pressurized and cannot balance out the force due to pressure acting at the inner face of the ring. The force due to the differential area being acted upon by the 3,000 psi oil can easily be calculated at 8,574 lb (38.137 kN).

Figure 10 illustrates the pressures acting on the outboard seal ring. The net force acting on the ring is the normal force existing between the seal ring and the seal housing. The sliding friction force required to move the seal ring radially under these conditions is calculated below using a coefficient of friction for lubricated steel equal to 0.11.

$$F_f = \mu N$$
$$F_f = 0.11 \times 8{,}574$$
$$F_f = 943 \text{ lb}$$

This friction force is 943 lb (4.19 kN) on each seal for a total of 1,886 lbs (8.39 kN) and opposes shaft radial motion. ►

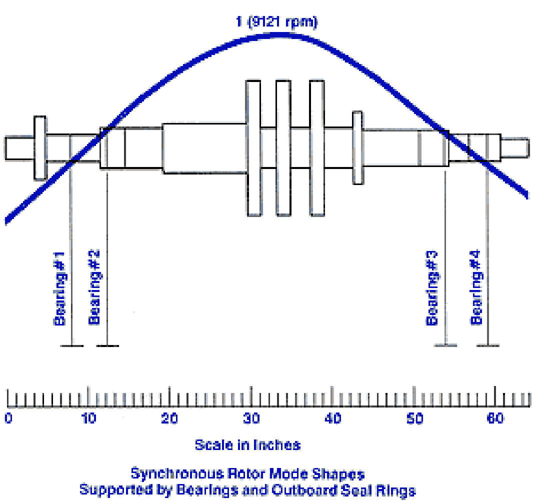

Figure 11
Baseline rotor model showing the effects of floating seal lockup

It is 5.9 times the rotor assembly weight of 320 lb (145 kg). Therefore, this seal friction force can support all or part of the rotor weight during operation.

The diametral clearance between the outer seal ring and the shaft is 0.0055 in (140 μm). This is very similar to the clearance of the two tilting pad bearings in this compressor. The amount of load carried by either of the outboard seal rings may depend upon their position with respect to the bearing when lockup occurs.

When the floating seals lock up, they behave like additional bearings on the rotor. This causes the rotor system stiffness to increase, elevating the balance resonance speed from the expected 5,700 rpm to approximately 9,100 rpm (Figure 11). The bearing stiffness used to model the seal behavior is 2.0×10^6 lb/in (3.58×10^8 g/cm). This was established by trial and error to achieve the 9,100 cpm balance resonance reported by the data. It is reasonable that the seal rings should be stiffer than the tilting pad bearings since the seal rings are full cylindrical bearings as well as externally-lubricated bearings, each being stiffer than a tilting pad bearing.

Recall that during the initial startup of this compressor, the balance resonance speed was estimated to be higher than 10,000 cpm. This further elevation in balance resonance speed is most likely due to labyrinth seal rubs as evidenced by the scored shaft when disassembled.

Conclusion

During twelve months of operation the compressor has continued to operate with the balance resonance very near the operating speed. There are no indications of seal wear or changes in the vibration amplitudes.

Subsequently, a variable speed test was performed increasing compressor rpm from 8,330 rpm to 11,000 rpm (Figure 12). This test verified that the balance resonance speed was in the range of 8,800 — 9,200 rpm. This speed range is also the normal operating range for the compressor. The compressor rotor is being supported by the outboard seal rings as well as the tilting pad bearings at each end of the rotor.

The seal rings can operate as a bearing with adequate fluid film to support the shaft weight, but they make poor candidates for bearings due to their random positioning and poor embeddability. Each compressor startup may position the rotor in a slightly different radial position with respect to the fixed case.

Continued operation may occur without incident except that premature seal ring wear is expected due to heavy radial loading. In similar applications, when oil seals influence the compressor rotordynamic response, gas seal technology has proven to be an alternative to consider. Dry gas seals use the process gas as a buffer gas that separates the stationary and rotating elements of a face seal. The plant is currently investigating the use of dry gas seals to avoid operating this compressor near the balance resonance speed. ■

References:

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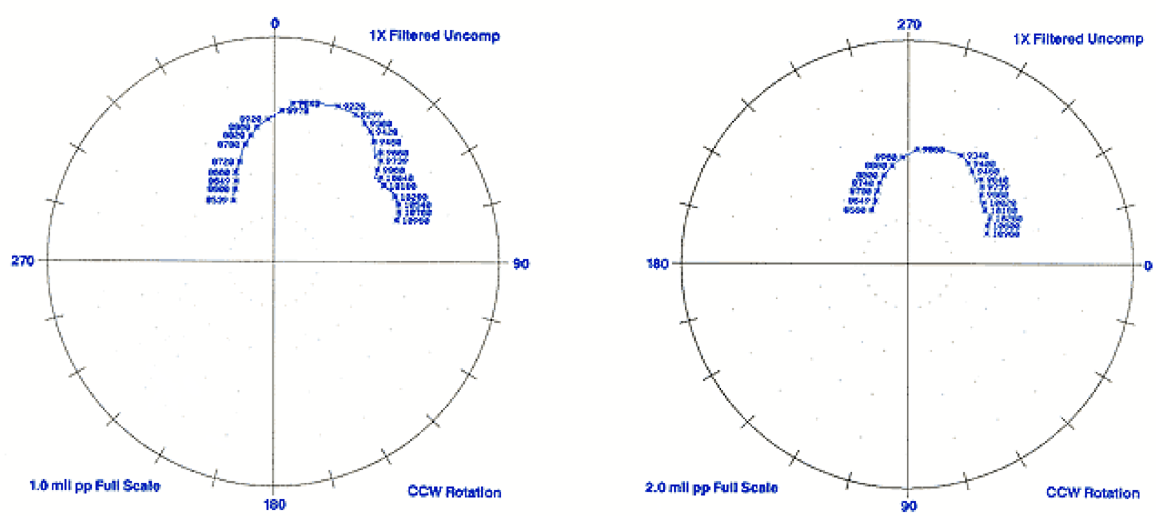

Figure 12
Inboard and Outboard compressor variable speed test results